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Lawrence B. Hanlon

Lawrence B. Hanlon

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1660 L STREET, N.W. • SUITE 613 • WASHINGTON, D.C. 20036 • TEL. (202) 296-1344

Valve

The invention relates to a valve, especially a proportional seat valve or gate valve, having a valve housing and at least three fluid ports extending through the valve housing, and with a main piston guided in the valve housing and a pilot valve which effects pilot control and which can be actuated by a magnet means which can carry current.

A generic valve is known from EP-A-0 893 607. This known valve is a magnetically operated drain valve in which, between a load pressure port (P) and a drain port (T) in the lifting module of a forklift, a seat closing element is assigned to the main valve seat and in the closing direction can be pressurized to a variable difference between the drain pressure and the control pressure derived from the load pressure, a pilot valve which can be actuated by a magnet means being provided with a pilot piston for the control pressure. The main valve which is formed by the main valve seat and the seat closing element is assigned a pressure compensator with a seat valve sealing function which with the main valve forms a two-way flow regulator which is independent of the load pressure and which is leak-proof under the load pressure in the closing position of the main valve.

This known approach discloses a structurally simple, magnetically operated drain valve of compact size, with which it is possible to implement a so-called ramp function independently of the load pressure. A ramp function is defined as the possibility of controlling the flow amount depending on lift and independently of pressure. But it has been shown that the known solution for lowering the load in hydraulic lifting devices does not meet the high demands as desired, specifically achieving a high no-load lowering speed with little leakage and precise metering this lowering speed.

Control devices for hydraulically operating lifting means are commercially available and they use directly controlled valves which however are not suitable for high volumetric flow due to the design, so that in general pilot-controlled valves are preferred. In so-called barometrically pilot-controlled valves an independent pressure supply which makes available the required pressure for adjusting the main piston is necessary. This pressure is generally 10 to 20 bars and is often produced by an external supply, for example the feed pump of the traveling mechanism, in a forklift with an internal combustion engine. In lifts with an electric drive however there is no pertinent external supply so that the required control pressure can only be taken from the load pressure. When lowering at no load the available control pressure can then drop to approximately 2 bars with the result that in no-load lowering the lowering process is hampered.

On the basis of this prior art, the object of the invention is to make available a valve which at low cost permits a high no-load lowering speed with few components in a reliable manner and allows precise metering of the lowering speed with simultaneously little leakage. This object is achieved by a valve with the features specified in claim 1 in its entirety.

In that, as specified in the characterizing part of claim 1, with the pilot control opened, fluid travels from one of the two ports which can be actuated by the main piston by way of a cross-sectional constriction in the main piston and the pilot control to the third port, which third port can

be actuated by the pilot piston and in that as a result of the accompanying pressure drop the main piston travels into a respective control position which can actuate the two fluid ports with respect to the amount of fluid, a pilot-controlled proportional seat valve or gate valve is formed which at a very low pilot pressure, for example < 2 bars, already completely opens and thus permits prompt no-load lowering.

If current is supplied to the magnet means to open the pilot control, the main piston is pushed up, the piston lift of the main piston being proportional to the magnet current. Since the position of the main piston accordingly always corresponds to the force of the magnet, a valve can thus be configured with which precise metering of the lowering speed is possible with simultaneously low leakage for the valve as claimed in the invention.

In one preferred embodiment of the valve as claimed in the invention, a compression spring is configured between the main piston and the pilot piston, the piston lift of the main piston with the pilot control opened being proportional to the magnet current of the magnet means. The compression spring which acts on the main piston reports the position of the main piston back to the pilot piston and consequently to the pilot control so that any disturbing variables, caused by flow forces, for example, can be directly adjusted and the position of the main piston accordingly corresponds to the applied magnet force. When no current is being supplied to the magnet means, flow through the valve is possible due to the compression springs of the two ports with the capacity to be controlled by the main piston as a spring-loaded return valve.

By preference here provision is made such that the compression spring engages a recess of the main piston into which the cross-sectional constriction in the form of an orifice discharges, on the free end of the compression spring which is facing the pilot piston there being a contact piece which is connected to the free end of the pilot piston by way of a contact ball. The latter permits unhampered operation and interaction of the pilot piston with the main piston.

In another embodiment of the valve as claimed in the invention there is preferably a selector valve in the main piston, the selector valve preferably having a cross-sectional constriction. In this version, in the absence of current, the valve can be blocked from one pressure port to another, which ports can be actuated by the main piston, and when current is supplied to the magnet means under the corresponding pressure conditions a volumetric flow between the fluid ports can thus be controlled. In one alternative embodiment the cross-sectional constriction of (choke or orifice) can also be located in a fluid-carrying channel downstream from the selector valve in the direction of the interior of the main piston.

In another preferred embodiment of the valve as claimed in the invention, the magnet means has at least one armature, a coil and a pole tube which is designed as of a pushing or pulling system, i.e., that the armature is moved out of or into the pole tube when the coil is supplied with current, and that when using a pulling system another compression spring moves the pilot piston in the direction of an opened pilot control. If the "pulling" pole tube is equipped with the additional compression spring mentioned in the foregoing which keeps the pilot piston in the open position, which corresponds to the fully energized state for the "pushing" pole tube, by switching the magnet means the pilot control and thus the valve can be completely closed. By replacing a "pushing" pole tube with a "pulling" pole tube, a valve which is open without current can therefore be formed from a proportional seat valve which is closed without current. If a pilot spring applies an adjustment force to the pilot piston, this is not absolutely necessary with respect to the operating property of the magnet system; but it improves the return of the pilot piston and thus the operating dynamics for the entire valve.

In another preferred embodiment of the valve as claimed in the invention, the pilot control is designed as a gate valve in which a pilot piston of cylindrical design at least on its free end is guided to be movable in the longitudinal direction into a corresponding elongated recess in parts of the valve housing. In this way uniform operating behavior is achieved even under the most varied

operating conditions, and by maintaining a sufficiently small sealing gap on the pilot piston the desired forklift tightness can be guaranteed.

In a different embodiment of the valve as claimed in the invention, by preference provision is made such that the pilot control is designed as a seat valve in which on the free end of the pilot piston there is a preferably cone-like closing and sealing part which interacts with a seat part, formed by parts of the valve housing. In this version as a seat valve the pilot control is free of leaks; but this entails the disadvantage that the pilot piston is no longer optimally pressure-equalized and is also subject to friction by the seal in its motion. If the pilot control is designed as a valve without a seal, the valve is no longer free of leaks, but possibly inhibitory friction in operation is thus largely precluded, which ensures that the valve performs its choke function. Preferably, to enhance the sealing on the outside circumference of the pilot piston, additional sealing parts may be provided.

The described valve is especially well-suited for all applications in which a large volumetric flow must be controlled with a low control pressure; this is often the case in the implementation of the lowering function in electric forklifts.

The proportional seat valve can generally be used as a proportional choke valve for very large volumetric flows. In order to keep Δ_p small at high volumetric flows, it may be necessary to enlarge the seat diameter in the valve body. The necessary control pressure for complete opening of the valve thus in fact increases; but it is always notably less than that of the known barometrically actuated valves.

In one preferred embodiment, the valve as claimed in the invention in a valve system serves the function of an adjustable metering orifice of a flow regulator in conjunction with a pressure compensator. In this configuration the flow amount can be controlled depending on the lift and

independently of the pressure (ramp function), and during lowering the volumetric flow to be managed can be limited in terms of its maximum; this serves to enhance reliability.

The valve design as claimed in the invention is detailed below in the drawings in which in diagrammatic form, not drawn to scale,

FIGS. 1 and 2 show a longitudinal section through two different embodiments of the proportional seat valve as claimed in the invention, viewed in the direction of looking at the figures the respective graphic symbol of the valve being shown at top left;

FIG. 3 in the form of an operating diagram shows the use of the valve as shown in FIG. 1 for a load lowering means for forklift units with maximum volumetric flow limitation and with load compensation;

FIG. 4 shows enlarged a longitudinal section through the lower part of another embodiment in the form of a proportional gate valve including its graphic symbol which is shown at top left.

The valve which is shown in FIG. 1 in a longitudinal section constitutes a so-called proportional seat value with a valve housing 10 which has seals and seal stacks on the outer circumferential side and which is designed as a screw-in cartridge for fixing the valve on other machine or vehicle parts for purposes of controlling a hydraulic circuit (not shown). Furthermore, the valve can also be designed as a kit. The valve housing 10 has three fluid ports 1, 2, 3, one fluid port 1 on the front engaging on the lower end of the valve housing 10 and the other two ports 2 and 3 being configured radially on the valve housing 10 on the outer circumferential side, the fluid port 2 at two different points 2a, 2b extending radially through the valve housing 10 and the third fluid

port 3 discharging by way of transverse holes 12 into the interior of the valve housing 10 which in this area has a valve insert 10a which is made with a screw-in bevel 14. In the valve housing 10 there is a main piston 18 which can move axially to the longitudinal axis 16 of the valve and which on its free end and adjacently opposite the fluid port 2a forms a seat valve 20 with the assigned wall parts of the valve housing, for this purpose the main piston 18 on its free end being provided with a conically extending valve surface 22. Next to the main piston 18 within the valve housing 10 a pilot piston 24 is guided in the longitudinal direction so as to be movable and is part of a pilot control designated as a whole as 26.

Viewed in the line of sight to FIG. 1, the valve housing 10 on its top end has a magnet means 28 which can carry current, with attachment plugs 30 for connection to an electrical means and for supplying current to a coil winding 32 which comprises an armature 34 which is mounted so as to be movable in the longitudinal direction within a profiled tube 36 and is used to actuate the pilot control 26, especially in the form of a pilot piston 24. This structure of a magnet means 28 is relatively well known in the prior art so that it is not detailed here.

According to the operating diagram as shown in FIG. 1 the main piston 18 is in its closed position, i.e., the seat valve 20 is blocking the fluid path between the fluid ports 1 and 2a. A cross-sectional constriction 38 located radially on the outer circumference of the main piston 18, preferably in the form of an orifice, discharges into a radial recess 40 of the main piston 18 which extends between the fluid port 2b and a radial projection 42 of the main piston 18 which separates the fluid port 2a from the radial recess 40. The main piston 18 is provided with a recess 44 into which the orifice 38 discharges and within this recess 44 which extends in the direction of the longitudinal axis 16 there is a compression spring 46 which with its one free end is in contact with the bottom of the recess 44 and with its other free end acts on a contact piece 48 which on its free end bears a contact ball 50 in a corresponding depression, on the top of which the free end of the pilot piston 24 is supported. In this way unhampered operation and actuation between the pilot

piston 24 and the main piston 18 is achieved, even in the event of possible tilting processes which can be equalized by the contact ball 50.

In the solution illustrated in FIG. 1 which viewed in the direction of looking at it is shown in terms of its operation in a conventional operating diagram at top left, in which the fluid ports 1, 2, and 3 shown there correspond to the ports as shown in the valve cross section, the pilot control 26 is designed as a gate valve in which the cylindrically configured pilot piston 24 at least on its free end is guided to be movable in the longitudinal direction in a corresponding longitudinal recess 42 which is circular in cross section in parts of the valve housing 10 in the form of a valve insert 10a. The pilot piston 24 on the outer circumferential side is conventionally enclosed by pressure relief grooves which at least partially ensure leak-tightness in this area of the pilot control 26. Between the underside of the valve insert 10a and the upper terminating end of the main piston 18 forming its back 54, the inner circumferential side of the valve housing 10 borders the control chamber 56 into which longitudinal channels 58, 60 of the valve insert 10a discharge, one longitudinal channel 58 with its other end discharging into an annular recess 62 of the pilot piston 24 and the other longitudinal channel 60 with its other free end discharging into an annular chamber 64 in which there is another compression spring 66 which is supported with its one lower end on the inner circumference of the valve insert 10a and with its other end on the radial widening 68 of the pilot piston 24, in the illustrated operating diagram as shown in FIG. 1 the radial recess 68 being supported with its outer flange on the front end of the magnet housing 70 which is inserted at this point in the valve insert 10a by way of a screw-in section. Furthermore, a radial annular channel 72 discharges into a radial chamber 74 between the inner circumferential side of the top end of the valve housing 10 and the outer circumferential side of the valve insert 10a in this area. In turn, the fluid port 3 discharges into this radial chamber 74 and on the opposite end in the illustrated operating position as shown in FIG. 1 the annular channel 72 is closed by the outside circumference of the pilot piston 24, with the actuated pilot piston 24 pressed down by the magnet means 28 when viewed in the direction of looking at FIG. 1, capable of establishing a fluid-carrying connection

between the control chamber 56, the longitudinal channel 58, the annular recess 62, the annular channel 72, the radial chamber 74, and the fluid port 3 by way of channel-like transverse holes 12.

For the sake of better understanding, at this point the proportional seat valve shown in FIG. 1, specifically intended for use in hydraulically operating lifting means, will be described in detail using a working example. If the magnet means 28 is supplied with current by way of the attachment plug 30, the armature 34 under the action of the field of the coil winding 32 migrates out of the pole tube 36 and in the process actuates the pilot piston 24 of the pilot control 26 against the action of the other compression spring 66 which with its reset force has the tendency to keep the radial widening 68 in contact with the lower end of the magnet housing 70. The indicated magnet force however in any case is sufficient to open the pilot control 26 against the action of the other compression spring 66 and the pilot oil flows from the load port 2 by way of the respective connecting point 2b into the radial recess 40 of the main piston 18 and from there by way of the cross-sectional constriction 38 (orifice) into the recess 44 of the main piston 18 in which the compression spring 46 is mounted. From there the pilot oil flows into the control chamber 56 and from there by way of the longitudinal channel 58, the annular recess 62 in the pilot piston 24 into the annular channel 72 and from there by way of the radial chamber 74 by way of the oblique holes 12 to the fluid port 3. In the process the pressure drops on the rear 54 of the main piston 18 and by the load pressure acting on the annular surface between the outside piston diameter and the valve seat diameter of the main piston 18 at the location of its seat valve 20 the main piston is pushed up against the action of the compression spring 46, viewed in the line of sight to FIG. 1. The pertinent piston lift of the main piston 18 is proportional to the magnetic current. The compression spring 46 located in the main piston 18 reports the position of the main piston 18 back to the pilot piston 24 so that disturbing variables such as for example the flow forces can be adjusted in this way and the position of the main piston 18 thus always corresponds to the magnetic force of the magnet means 28 in the current-carrying state. Without current the main piston 18 assumes its position shown in FIG. 1 and

in this position as a result of the compression spring 46 the valve acts like a spring-loaded return valve 76 relative to the control of possible fluid flow between the fluid ports 1 and 2.

With this configuration a pilot-controlled proportional seat valve is implemented which at a very low pilot pressure (for example < 2 bars) already completely opens and thus permits rapid no-load lowering so that its use is of interest especially in electrically operated forklifts which do not have an external supply which is necessary to ensure the required pressure for setting the main piston in barometrically pilot-controlled valves, as they are known in the prior art.

The pilot spring in the form of the other compression spring 66 is not absolutely necessary, but, as already described, it improves the return of the pilot piston 24 and thus the dynamics of the valve as a whole. The pilot control 26 implemented in FIG. 1 is designed as a gate valve, this being the best solution for uniform operating behavior under different operating conditions, but this is accompanied by the disadvantage that the valve shown in FIG. 1 consequently is subject to leakage. By maintaining a sufficiently small sealing gap on the pilot piston 24 however the desired forklift tightness can be ensured.

The pole tube 26 used in FIG. 1 is designed as a so-called pushing system in which the armature 34 emerges from the pole tube 36 when the coil winding 32 is supplied with current. In "pulling" systems, that is in a so-called "pulling" pole tube however the armature 34 moves into the pole tube 36. If the "pulling" pole tube is equipped with a compression spring (not shown) which keeps the pilot piston 24 in the open position, which corresponds to the state of full current supply for the pushing pole tube 36, by switching the magnet means 28 the pilot control 26 and thus the valve can be completely closed. By replacing a "pushing " pole tube 36 by a "pulling" pole tube, a valve which is open without current can thus easily be configured from a proportional seat valve which is closed without current, if the requirements of practical application make this necessary.

FIG. 3 shows one example of application of the proportional seat valve shown in FIG. 1 for a hydraulically operating lifting means designated as a whole as 78. The hydraulic lifting means 78 has a load fork 80 of conventional design which can be raised and lowered by way of an actuator cylinder 82. For the sake of clarity of illustration, the behavior of the lifting frame of the lifting means 78 is shown here as a choke 84 in terms of its hydraulic behavior. Moreover the piston side of the actuator cylinder 82 can be connected to the tank T by way of the connecting line 86. The symbolically shown pressure gauges with designations P_H, P_2, P_1, and P_T within the scope of a test set-up would permit tapping of pressure valves in individual travel positions of the lifting means 78 within the connecting line 86. As FIG. 3 furthermore shows, a known pressure compensator 90 with a choke function is connected to the connecting line 86 and it is controlled by the prevailing pressure in the connecting line 86 by way of the connecting point 92. In this way, as shown in FIG. 3 a valve system is implemented consisting of a valve as shown in FIG. 1 and in conjunction with the known pressure compensator 90 an adjustable metering orifice of a flow regulator is implemented. The proportional seat valve as shown in FIG. 1 can be used in this way as a proportional choke valve for very large volumetric flows and with the illustrated valve system solution as shown in FIG. 3 the maximum volumetric flow can be limited when the load fork 80 is being lowered (with or without a load); this benefits reliability during operation of the lifting means. In particular, with this solution at a low control pressure a large volumetric flow can be controlled.

The embodiment as shown in FIG. 2 constitutes a version of the embodiment shown in FIG. 1 and accordingly is only explained to the extent it differs significantly from the embodiment in FIG. 1. In this respect the same reference numbers as in FIG. 1 are used for the same parts and what has been stated previously then also applies in this respect to the modified embodiment as shown in FIG. 2.

In the embodiment as shown in FIG. 2, on the lower front end of the main piston 18 there is a selector valve 95, the selector valve 95 having a cross-sectional constriction. The pertinent orifice

function is present twice in two throughflow directions from 1 to 2 and vice versa relative to the fluid ports. The selector valve 95 has a valve ball 98 which can be moved in a transverse channel 96 and which, depending on the incident fluid flow direction from the fluid port 1 to 2 or vice versa, on the one hand blocks the fluid connection point of one selector valve insert 95a and of the other selector valve insert 95b with their respective cross-sectional constriction 38. The indicated transverse channel 96 in the longitudinal direction of the valve has a longitudinal channel 100 which discharges into the recess 44 in the main piston 18 with the compression spring 46. In the embodiment as shown in FIG. 2, the pilot control 26 is designed as a seat valve with a seal. For this purpose the pilot piston 24 on its bottom free end has a cone-like closing and sealing part 102 which interacts with the seat part 104 on the bottom end of the valve insert 10a. Instead of the longitudinal channel 58, the modified solution as shown in FIG. 2 in the pilot piston 24 has transverse channels 106 which are furthermore connected to one another to carry fluid by way of a central longitudinal channel 108 so that in this way with the pilot control 26 opened the fluid flow from the fluid port 2 to the fluid port 3 is ensured. Furthermore, the pilot piston 24 on the outer circumferential side has a sealing system 110 within the annular chamber 64. In the illustrated version as a seat valve as shown in FIG. 2, the pilot control 26 is also free of leaks, the pilot piston 24 no longer being optimally pressure-equalized, but rather is also made subject to friction by the sealing system 110. If the indicated seal were omitted, the disadvantage of friction would not arise. But then this valve would no longer be free of leaks.

With this valve as shown in FIGS. 1 and 2, high no-load lowering speeds can be achieved in hydraulic lifting means with simultaneously precise metering of the lowering speed and with little leakage.

FIG. 4 relates to a modified valve embodiment compared to the illustrated versions in FIGS. 1 and 2. Thus FIG. 4 relates to the lower valve part which is designed as a gate valve, especially a proportional gate valve. Instead of the previously described conical valve seat 20, the free end of the

main piston 18 is made cylindrical, and is guided in a cylindrical inner circumferential surface of the lower end of the valve housing 10. With the main piston 18 raised, in this way the fluid-carrying choked connection between the valve port 2a and the free fluid entry side is established by way of the fluid-carrying part 112 on the front end of the valve housing 10. The corresponding operating diagram is shown at top left viewed in the line of sight to FIG. 4, the pilot control for this valve version which is designed as a gate valve being executed accordingly, as described in the foregoing for the valve versions as shown in FIGS. 1 and 2.